2022

Compressor Design Vs. Refrigerants Properties: What Affects Compressor Efficiency More?

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Compressor design vs. refrigerants properties: What is the main influence on compressor efficiency?

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ABSTRACT

Many knowledge gaps exist on the correlation between compressor efficiency, compressor design, and refrigerant properties. However, particularly for refrigerant assessment and selection, knowledge about compressor behavior is essential as the entire cycle efficiency substantially depends on compressor efficiency. Most selection approaches are based on simulations using process models to assess the performance of refrigerants. The process models applied are mostly simple cycle calculations using a fixed or pressure-dependent isentropic efficiency to represent compressor behavior. Usually, isentropic efficiency is assumed to be constant for all considered refrigerants. However, experimental data of positive displacement compressors imply that compressor efficiency depends more on the refrigerant than on the compressor used. This study aims to analyze for reciprocating piston compressors whether tailoring compressors to refrigerants can overcome the differences in isentropic efficiency caused by refrigerant properties. Using a loss-based model of reciprocating piston compressors that is fitted and validated on experimental data, compressor tailoring to refrigerants is simulated by adjusting the displacement and reducing losses from specific loss mechanisms. The study shows that isentropic efficiency is substantially refrigerant-dependent and less compressor geometry-dependent. Overcoming this refrigerant dependency by tailoring the compressor to each refrigerant is limited. Reducing losses from a specific loss mechanism can lead to a higher efficiency increase for refrigerants more affected by this loss mechanism. However, this effect is not sufficient to equalize the isentropic efficiency of all refrigerants or change the fundamental refrigerant ranking.

1. INTRODUCTION

The global demand for heat pumps and cooling process is rising: Heat pumps are regarded as a key technology for net-zero energy systems; cooling demand is increasing due to global warming and an increasing standard of living. However, both heat pumps and cooling processes currently employ refrigerants (mostly hydrofluorocarbons, HFCs) with high global warming potential (GWP). Thus, alternative refrigerants with low GWP need to be identified for vapor-compression heat pumps and cooling processes. Additionally, high-efficient refrigerants must be selected for emerging applications such as high-temperature heat pumps.

Today, refrigerants are assessed and selected based on process simulations. The selection methods obtain an optimal combination of refrigerant (-mixture) and process and have widely replaced methods that rely on heuristics. Methods that use process simulations can be divided into screening approaches (Bell et al., 2019; McLinden et al., 2017; Saleh et al., 2020) and integrated design approaches (Frutiger et al., 2018; Roskosch & Atakan, 2015; Schilling et al., 2017). In screening approaches, a fluid database is screened based on target refrigerant properties and process optimizations. Integrated design approaches solve the inverse problem and optimize refrigerant properties (Frutiger et al., 2018;

Roskosch & Atakan, 2015) or molecular structures (Kuprasertwong et al., 2021; Schilling et al., 2017) together with the process.

All these methods require a process model that accounts for the impact of the refrigerant. In this context, compressor modeling is particularly crucial. The isentropic compressor efficiency was shown to significantly influence the overall thermodynamic performance of the process (Roskosch et al., 2021; Sánchez et al., 2017). Furthermore, studies analyzing the performance of compressors show that the isentropic efficiency strongly depends on the refrigerant used (Sánchez et al., 2017; Sánchez et al., 2018; Venzik et al., 2017). Still, the process models applied for refrigerant selection usually assume constant compressor efficiencies for all refrigerants (Bell et al., 2019; Frutiger et al., 2018; Kuprasertwong et al., 2021; Roskosch & Atakan, 2015; Saleh et al., 2020) or use polynomials mainly accounting for pressure-ratio-dependent isentropic efficiencies (McLinden et al., 2017). Frequently, the assumption is made that, although the efficiency of a given compressor strongly depends on the refrigerant, similar efficiencies can be achieved for all refrigerants when the compressor design is tailored to each refrigerant.

The objective of this study is to verify this assumption. For this purpose, a loss-based model is derived for reciprocating piston compressors to calculate refrigerant-dependent compressor efficiencies (Section 2.1). The model accounts for differences in compressor efficiencies of refrigerants at constant operating conditions of the overall process in a vapor-compression heat pump. The compressor model considers friction, flow, and electric losses and therefore, covers the main compressor losses. These losses are described by fitting parameters and equations that correlate the losses with refrigerant properties and the compressor geometry (Section 2.2). The model parameters are fitted to a data set taken from a compressor manufacturer. Subsequently, the compressor model is validated on three additional experimental data sets (Section 3).

To analyze whether isentropic efficiency of positive displacement compressors is mainly an issue of compressor design, we tailor the compressor to individual refrigerants (Section 4). Generally, the compressor design can be tailored in two ways to a refrigerant: changing the displacement and reducing the loss mechanism crucial for the considered refrigerant. Tailoring the displacement is a standard measure to achieve a similar heating power of the vapor-compression process. Refrigerants with small densities usually require a larger compressor displacement. Reducing a specific loss mechanism is an option to alleviate differences between refrigerants in isentropic efficiency. For example, refrigerants with larger densities have larger mass flow rates which usually cause higher flow losses. The isentropic efficiency of these refrigerants can particularly benefit when special attention is paid to optimizing the flow geometry. We analyze the influence of both tailoring options on the isentropic efficiency for various refrigerants.

Our study shows that compressor efficiency mainly depends on the refrigerant properties and less on the geometry. Tailoring the compressor to refrigerants by reducing losses of a specific loss mechanism can change the ranking of neighboring refrigerants and slightly converge the efficiencies of all refrigerants. However, realistic measures for tailoring compressors to refrigerants do not have the potential to overcome differences in compressor efficiency caused by refrigerant properties. Equal compressor efficiencies for all refrigerants or changes in the fundamental refrigerant ranking might not be achievable for reciprocating piston compressors.

2. Modeling

This section is divided into two subsections: In Section 2.1, a loss-based compressor model is derived that accounts for the refrigerant dependency of a given compressor. Section 2.2 shows how the compressor model can be used to determine the compressor geometry needed to achieve a desired heating power of a heat pump cycle.

2.1 A loss-based compressor model

The compressor model derived here particularly accounts for the refrigerant dependency of two-cylinder reciprocating piston compressors. The compressor is initially considered to have a fixed geometry and is part of a vapor-compression heat pump cycle working at constant operating conditions. The conditions of the heat pump cycle are defined by the evaporation and condensation temperatures, the superheating at the compressor inlet, and the rotational speed of the compressor.

Generally, the thermodynamic behavior of a compressor is described by two efficiencies: The isentropic and the volumetric efficiency. One definition of the isentropic efficiency η_{is} relates the isentropic compressor power P_{is} to the electric power consumption P_{elec}.

$$\eta_{\mathrm{is}} = \frac{P_{\mathrm{is}}}{P_{\mathrm{elec}}} \tag{1}$$

The volumetric efficiency η_{vol} relates the actual volume $\dot{V}$ (or mass) flow rate to the theoretical value $\dot{V}_{\mathrm{theoretical}}$ which depends on the cylinder displacement V_{cyl}, the number of cylinders n_{cyl} and the mechanical rotational speed f_{mech}.

$$\eta_{\mathrm{vol}} = \frac{\dot{V}}{\dot{V}_{\mathrm{theoretical}}} = \frac{\dot{V}}{V_{\mathrm{cyl}} \cdot n_{\mathrm{cyl}} \cdot f_{\mathrm{mech}}} = \frac{\dot{m}}{V_{\mathrm{cyl}} \cdot n_{\mathrm{cyl}} \cdot f_{\mathrm{mech}} \cdot \rho_{\mathrm{in}}} \tag{2}$$

The electric power consumption can be split into the isentropic compressor power and the cumulative compressor losses P_{loss}.

$$P_{\mathrm{elec}} = P_{\mathrm{is}} + P_{\mathrm{loss}} \tag{3}$$

Compressor losses occur at various places in a piston compressor and are caused by multiple mechanisms. According to (Phillippi, 2016), compressor losses are divided into the following mechanisms:

- friction of bearings, piston rings, and other rubbing parts,
- electrical losses,
- flow losses of suction and discharge, mainly in valves,
- pressure losses due to valve opening,
- re-expansion caused by a fixed clearance volume,
- refrigerant leakage,
- driving auxiliary aggregates, e.g., oil pumps.

The magnitude of each loss mechanism generally depends on various factors (e.g., construction, operating conditions, and refrigerant properties), and, in turn, each loss affects the compressor efficiencies differently. For example, the re-expansion of the compressed gas, caused by the fixed clearance volume, directly decreases the refrigerant mass flow rate since the suction gas can only enter the cylinder when the gas remaining in the clearance volume has been re-expanded below the suction pressure. As a first-order effect, the reduced mass flow rate directly reduces the volumetric efficiency. However, second-order effects also occur. For example, the re-expanded gas is due to the irreversibilities of the previous steps (compression, rejection and re-expansion) at a higher temperature than the suction gas, increasing the compression work.

The compressor efficiency model derived here only focuses on the first-order effects of the most crucial loss mechanisms. These mechanisms are friction losses $P_{loss,fric}$, flow losses $P_{loss,flow}$, electrical losses $P_{loss,elec}$, and heat transfer losses $P_{loss,ht}$ (Hundy, 2016).

$$P_{\mathrm{loss}} = P_{\mathrm{loss,fric}} + P_{\mathrm{loss,flow}} + P_{\mathrm{loss,elec}} + P_{\mathrm{loss,ht}} \tag{4}$$

Friction happens at various places within a compressor. The magnitude of friction losses typically depends on the construction, the lubrication, and the rotational speed. The lubrication properties of the oil are affected by temperature, the amount of dissolved refrigerant, and the oil's properties (Da Riva & Del Col, 2011). Since compressor manufacturers prescribe different oils specified to the refrigerant used, we assume that lubrication properties are ultimately similar for common refrigerants and neglect differences in lubrication in our model. Therefore, friction losses are approximately constant for a given compressor under similar operating conditions and can be described by a constant value a.

$$P_{\mathrm{loss,fric}} = \mathrm{a} \tag{5}$$

Flow losses occur along the entire route of the gas from the suction to the discharge line. However, flow losses are particularly large in the valves due to small openings and flow deflection (Phillippi, 2016). A frequently used approach to calculating flow losses in reciprocating piston compressors is assuming an incompressible fluid (Dutra & Deschamps, 2015; Roskosch et al., 2017; Stouffs et al., 2001). According to the Darcy-Weissbach equation, a pressure drop is defined by:

$$|\Delta p| = b^* \cdot \rho \cdot u^2 = \frac{b^*}{\mathrm{A}_{\mathrm{flow}}^2} \cdot \frac{\dot{m}^2}{\rho} = b \cdot \frac{\dot{m}^2}{\rho} \tag{6}$$

where u is the flow velocity, ρ is the density and $b = b^*/A_{\text{flow}}^2$ is a proportionality factor. For an incompressible fluid, the reversible power needed to compensate for a pressure drop Δp is derived from:

$$P_{\Delta p} = \dot{m} \cdot \int_{p_1}^{p_1+|\Delta p|} \frac{1}{\rho}\,\mathrm{d}p = \dot{m} \cdot \frac{|\Delta p|}{\rho} \tag{7}$$

From equations (6) and (7), under assuming $\rho = \rho_{\text{in}}$, follows:

$$P_{\text{loss,flow}} = b \cdot \frac{\dot{m}^3}{\rho_{\text{in}}^2} \tag{8}$$

This equation includes many simplifications: Usually, the flow is not incompressible and factor b includes, besides geometrical properties, refrigerant properties such as viscosity and compressibility. However, in particular, the differences in viscosity and compressibility of usual refrigerants at comparable conditions are smaller compared to changes in the density. For example, an analysis of ten common refrigerants from different substance groups (natural refrigerants, HFOs, and HFCs) at T = 15 °C and $p = p_{\text{evap}}(0°\text{C})$ shows relative standard deviations of 16% and 2% for viscosity and compressibility but 37% for the density (Lemmon et al., 2018). Since the density is already considered by equation (8), we decided for simplification to neglect the influence of refrigerant properties on b.

Electrical losses are mainly caused by Joule heating in the windings of the electric motor (Dutra & Deschamps, 2015). The magnitude of electrical losses caused by Joule heating depends on the motor efficiency and the power consumption.

$$P_{\text{loss,elec}} = (1 - \eta_{\text{motor}}) \cdot P_{\text{elec}} \tag{9}$$

However, the motor efficiency usually varies only by a few percentage points over the designed load range of the motor (Dutra & Deschamps, 2015). Therefore, the electrical losses are assumed to be proportional to the electric power consumption of the compressor with a constant factor c = 1 - η_{motor}.

$$P_{\text{loss,elec}} = \mathrm{c} \cdot P_{\text{elec}} \tag{10}$$

First-order heat transfer losses are heat losses to the environment. Heat losses from the electric motor and the mechanical parts due to friction are already considered in $P_{\text{loss,fric}}$ and $P_{\text{loss,elec}}$. A further heat loss to the environment comes from the in-cylinder gas. However, the amount of transferred heat does not significantly change with the refrigerant used since mean gas temperatures are similar for akin compressor operating conditions which are considered here. We assume that a compressor-specific amount of heat losses is already covered by parameter a, and thus, set $P_{\text{loss,ht}}$ to zero.

Combining all loss terms leads to:

$$P_{\text{loss}} = a + b \cdot \frac{\dot{m}^3}{\rho_{\text{in}}^2} + \mathrm{c} \cdot P_{\text{elec}} \tag{11}$$

From combining with equation (3) follows:

$$P_{\text{elec}} = \left(P_{\text{is}} + a + b \cdot \frac{\dot{m}^3}{\rho_{\text{in}}^2}\right) \cdot \frac{1}{1 - c} \tag{12}$$

The isentropic compressor power is defined by:

$$P_{\text{is}} = \dot{m}\left(h_{\text{out,is}} - h_{\text{in}}\right) \tag{13}$$

and the mass flow rate can be derived from equation (2):

$$\dot{m} = V_{\text{cyl}} \cdot n_{\text{cyl}} \cdot f_{\text{mech}} \cdot \rho_{\text{in}} \cdot \eta_{\text{vol}} \tag{14}$$

For the energetic compressor efficiency finally follows:

$$\eta_{\text{ener}} = \frac{V_{\text{cyl}} \cdot n_{\text{cyl}} \cdot f_{\text{mech}} \cdot \rho_{\text{in}} \cdot \eta_{\text{vol}} \cdot \left(h_{\text{out,is}} - h_{\text{in}}\right)}{V_{\text{cyl}} \cdot n_{\text{cyl}} \cdot f_{\text{mech}} \cdot \rho_{\text{in}} \cdot \eta_{\text{vol}} \cdot \left(h_{\text{out,is}} - h_{\text{in}}\right) + a + b \cdot \rho_{\text{in}} \cdot (V_{\text{cyl}} \cdot n_{\text{cyl}} \cdot f_{\text{mech}} \cdot \eta_{\text{vol}})^3}(1 - c) \tag{15}$$

The main drivers of reducing the volume flow rate of compressors compared to the theoretical value are the re-expansion caused by the dead volume and over compression. Fresh gas can only be sucked in again when the remining gas in the cylinder is re-expanded below the pressure of the suction line. By assuming ideal gas, an isentropic re-expansion and a specific dead volume V_{dead}, the effective suction volume V_{eff} can be calculated by:

$$V_{\text{eff}} = V_{\text{cyl}} - V_{\text{dead}} \cdot \left(\left(\frac{p_{\text{out}}}{p_{\text{in}}} \right)^{\frac{c_v}{c_p}} - 1 \right) \tag{16}$$

To consider over compression, the outlet pressure is increased by $k_2 \cdot \rho_{in}$. Here, we assume that high-density refrigerants cause larger over compression since larger mass flow rates have to flow through the valves.

$$V_{\text{eff}} = V_{\text{cyl}} - V_{\text{dead}} \cdot \left(\left(\frac{p_{\text{out}} + k_2 \cdot \rho_{\text{in}}}{p_{\text{in}}} \right)^{\frac{c_v}{c_p}} - 1 \right) \tag{17}$$

Inserting equation (17) into (2) and replacing the height of the dead volume by the fitting parameter k_1 leads finally to an expression for the volumetric efficiency:

$$\eta_{\text{vol}} = 1 - \frac{k_1}{H} \cdot \left(\left(\frac{p_{\text{out}} + k_2 \cdot \rho_{\text{in}}}{p_{\text{in}}} \right)^{\frac{c_v}{c_p}} - 1 \right) \tag{18}$$

Finally, the model derived has three compressor-specific parameters for the energetic efficiency (a, b, c) and two parameters for the volumetric efficiency (k_1 and k_2) that can be fitted to measured data. Although the model explicitly considers only the most crucial first-order loss mechanisms, it is assumed that fitting the parameters to measured data also partially covers other loss mechanisms and second-order effects.

2.2 Process- and refrigerant-specific compressor modeling

In heat pumps, different refrigerants lead to a strongly different heating power when the compressor remains unchanged (Roskosch et al., 2019; Venzik et al., 2017). The deviation in heating power is mainly due to large differences in refrigerant density. To achieve a similar heating power, the compressor is usually specifically selected by its displacement for the refrigerant used. Many compressor losses depend on the geometry and thus on the displacement. Therefore, a comparison of compressor efficiencies for different refrigerants should rely on different compressor geometries tailored to the refrigerant and the application. For heat pumps, tailoring to the application means achieving an equal heating power for all refrigerants.

When isobaric heat transfer and full condensation are assumed, the heating power $\dot{Q}_{\text{H}}$ of a heat pump cycle is defined by:

$$\left|\dot{Q}_{\text{H}}\right| = \dot{V}_{\text{theoretical}} \cdot \rho_{\text{in}} \cdot \eta_{\text{vol}} \cdot \left(h_{\text{in}} + \frac{w_{\text{is}}}{\eta_{\text{ener}}} - h(p_{\text{out}}, x = 0) \right) \tag{19}$$

According to equations (15) and (18), the heating power for a given refrigerant at defined operating conditions of the heat pump is, therefore, compressor-specific and depends on the theoretical volume flow rate $\dot{V}_{\text{theoretical}}$ and the loss parameters a, b, c and k_1, k_2. According to equation (2), the theoretical volume flow rate depends on the cylinder displacement, the number of cylinders, and the rotational speed. In relation to the compressor considered as the reference (REF, DS1), only compressors with two cylinders are considered in the following. The theoretical volume flow rate is then only compressor-specific by the cylinder displacement, defined by bore *D* and stroke *H*.

$$V_{\text{cyl}} = \frac{\pi}{4} \cdot D^2 \cdot H \tag{20}$$

For our model we assume that the fitting parameters of the volumetric efficiency (k_1 and k_2) do not depend on the compressor used. However, the volumetric compressor efficiency is still compressor-specific by the stroke *H* (equation (18)). Furthermore, the efficiencies of electric motors are usually very similar and do not depend on the specific motor used. Therefore, it is assumed that c is constant for all compressor models analyzed within this work. However, friction (a) and flow (b) losses strongly depend on the compressor geometry. To cover this effect, we use two equations correlating a and b with the cylinder geometry based on the compressor used for the fitting. In the following we use the index “ref” for the compressor used for fitting and the index “scale” when a and b are scaled by geometry (equations (21) and (22)). Friction losses are subjected to two mechanisms: One share of the entire friction losses is independent of the cylinder geometry (e.g., shaft bearings), and the other share depends on the piston diameter *D* and the stroke *H* as they define the friction surface. We assume a weighting of 50:50 as it turned out to be reasonable.

$$\frac{a_{\text{scale}}}{a_{\text{ref}}} = 0.5 + 0.5 \cdot \frac{D}{D_{\text{ref}}} \cdot \frac{H}{H_{\text{ref}}} \tag{21}$$

The amount of flow losses mainly depends on the valve orifice, which is limited by the cross-sectional area of the piston. Therefore, according to equation (6) (A_{flow} scaled by cross-sectional area), we assume:

$$\frac{b_{\text{scale}}}{b_{\text{ref}}} = \frac{D_{\text{ref}}^4}{D^4} \tag{22}$$

The ratio of D and H is usually in a small range close to 1. For example, an analysis of compressors with two cylinders showed values of D/H between 0.9 and 1.5 (BITZER Kühlmaschinenbau GmbH). For simplification, we assume D/H = 1.25 in the design. However, the influence of this ratio is discussed in the results section.

Combining equations (19) - (22) and equations (13) - (18) yields a set of equations enabling iteration of the necessary bore to obtain the target heating power $\dot{Q}_{\text{H,target}}$.

$$\left|\dot{Q}_{\text{H}}(D, D/H, A, B)\right| - \dot{Q}_{\text{H,target}} = 0 \qquad \dot{Q}_{\text{H,target}} > 0 \tag{23}$$

The vectors A and B account for refrigerant properties and the operating conditions of the heat pump, respectively. The loss-based compressor model can be used with any fluid property model. We use in the following REFPROP Version 10 (Lemmon et al., 2018) to calculate enthalpies, entropies, and the vapor-liquid-equilibrium.

3. Model fitting and validation

For model fitting and validation, four data sets are used (DS1 to DS4). The data sets DS1 (BITZER Kühlmaschinenbau GmbH), DS2 (Venzik, 2019) and DS3 (Sánchez et al., 2017) are taken from the literature, and the data set DS4 was produced by two of the authors and is already partly published (Arpagaus et al., 2018; Arpagaus & Bertsch, 2019). Each data set was obtained by a heat pump cycle operating under similar conditions (evaporation T_{evap}, condensation T_{cond} temperatures, compressor inlet T_{in} temperature, and rotational speed) but using various refrigerants. The heat pump cycles of DS1-4 were equipped with reciprocating piston compressors of different specifications (Table 1).

The model parameters (a, b, c, k_1, k_2) are fitted to data set DS1, which is considered to be the reference case (index: ref). The fitting by a least squares algorithm results in:

$a_{\text{ref}} = 91.59$ W

$b_{\text{ref}} = 12.11 \cdot 10^6\ \text{W} \cdot \text{s}^3 \cdot \text{m}^{-6}$

$c_{\text{ref}} = 0.062$

$k_{1,\text{ref}} = 3.49$ mm

$k_{2,\text{ref}} = 7.19 \cdot 10^3\ \text{m}^2 \cdot \text{s}^{-2}$

Table 1: Compressor data associated with the data sets (DS) used.

	DS1	DS2	DS3	DS4
Model	Bitzer, various	GEA-Bock, HG-HC-12P	n/a	Bitzer, 2DES-3Y
Type	Semi-hermetic	semi-hermetic	hermetic	semi-hermetic
No. of cylinders n_{cyl}	2	2	1	2
Bore D / Stroke H	30 mm / 33 mm	34 mm / 34 mm	n/a, assumption: D/H = 1.25	50 mm / 39.3 mm
$\dot{V}_{\text{theoretical}}$ at f_{elec} = 50 Hz	4.06 $\text{m}^3 \cdot \text{h}^{-1}$	5.40 $\text{m}^3 \cdot \text{h}^{-1}$	2.10 $\text{m}^3 \cdot \text{h}^{-1}$	13.40 $\text{m}^3 \cdot \text{h}^{-1}$
Data source	(BITZER Kühlmaschinenbau GmbH)	(Venzik, 2019)	(Sánchez et al., 2017)	(Arpagaus & Bertsch, 2019)

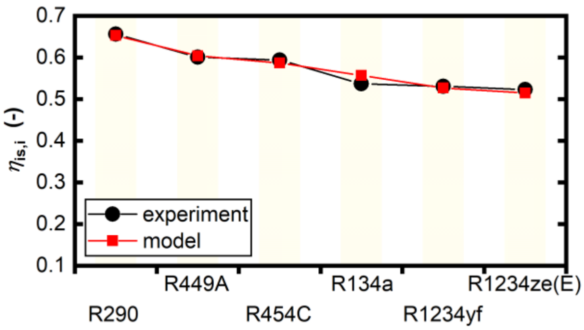

Figure 1: Calculated (squares) and measured (circles) isentropic compressor efficiencies for DS1. Calculated using fitted a_{ref}, b_{ref}, c_{ref}, $k_{1,ref}$, $k_{2,ref}$. T_{evap} = 0 °C, T_{cond} = 35 °C, T_{in} = 20 °C.

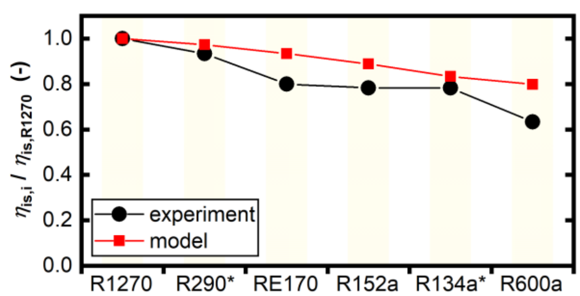

Figure 2: Calculated (squares) and measured (circles) rel. isentropic compressor efficiencies for DS2. Calculated using a_{scale}, b_{scale}, c_{ref}, $k_{1,ref}$, $k_{2,ref}$ and the respective measured inlet and outlet states. T_{evap} ≈ 0 °C, T_{cond} ≈ 33 °C, T_{in} ≈ 17 °C.

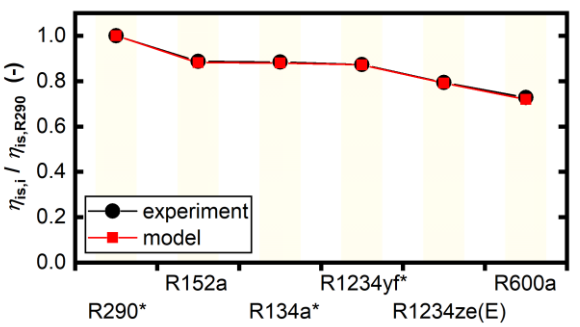

Figure 3: Calculated (squares) and measured (circles) rel. isentropic compressor efficiencies for DS3. Calculated using a_{scale}, b_{scale}, c_{ref}, $k_{1,ref}$, $k_{2,ref}$ and the respective measured inlet and outlet states. T_{evap} ≈ 0 °C, T_{cond} ≈ 35 °C, T_{in} ≈ 12 °C.

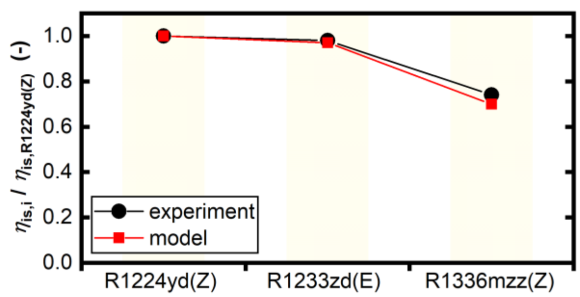

Figure 4: Calculated (squares) and measured (circles) rel. isentropic compressor efficiencies for DS4. Calculated using a_{scale}, b_{scale}, c_{ref}, $k_{1,ref}$, $k_{2,ref}$ and the respective measured inlet and outlet states. T_{evap} ≈ 25 °C, T_{cond} ≈ 70 °C, T_{in} ≈ 50 °C.

Using these fitting parameters, the compressor efficiencies calculated by the model show a good alignment with the measured values (Figure 1). For validation, the model is applied to the data sets DS2 – DS4 using the respective compressor geometry and operating conditions but the loss parameters resulting from the fitting to DS1 scaled by the respective geometry. The measured and calculated energetic compressor efficiencies (Figure 2 - Figure 4) are here normalized to the respective maximum value as we only aim to analyze and predict differences in compressor efficiencies due to refrigerant properties. The model results show a good alignment with the measured values also for the data sets DS2 – DS4, although the compressor geometry, the considered refrigerants and the operating conditions are extrapolated related to the data set used for fitting (DS1). The largest deviations are observed for DS2 (Figure 2). However, the model correctly predicts the refrigerant ranking which is particularly important for our study. Regarding DS2, we were not able to clarify whether the measurements or the model is the origin of the deviations. In summary, the validation shows that the model accurately reproduces the refrigerant ranking also for compressors, refrigerants, and operating conditions not considered in the fitting. Therefore, from a mathematical perspective, the model is appropriate to describe the refrigerant dependency of compressor efficiencies.

4. Results and discussion

To still achieve a constant heating power for various refrigerants, compressors with a different displacement are usually selected. We cover this aspect in our study by determining the required displacement to achieve a heating power of 3 kW, using the procedure described in Section 2.2. The loss parameters a_{ref} and b_{ref} are correlated with the compressor geometry according to equations (21) and (22)while c_{ref}, $k_{1,ref}$ and k_{2ref} are unchanged. For the ratio of bore and stroke D/H = 1.25 is assumed.

Substantially different compressor displacements for the various refrigerants (Figure 5 bottom, pentagons) result from the calculations. The biggest difference is observed between R1234ze(E) and R32: the required displacement to achieve a heating power of 3 kW is about three times higher for R1234ze(E). The difference is caused by differences in density (Figure 5 bottom, arrows). Refrigerants with higher densities require a smaller displacement to achieve the desired heating power. The isentropic efficiencies obtained for compressors tailored to each refrigerant by geometry (Figure 5 top, circles) show large differences for the various refrigerants. While the best performing refrigerant (R32) has an efficiency of 74%, the lowest isentropic efficiency is 58% (R1234ze(E)). It is observed that refrigerants which require smaller displacements tend to have larger efficiencies. Decreasing the compressor displacement by decreasing bore and stroke also decreases the friction losses ($a_{scale,R1234ze(E)}$ = 107.8 W, $a_{scale,R32}$ = 75.6 W) while the isentropic power is approximately constant (*Figure 5* bottom, triangles). This causes an increase in isentropic efficiency according to equations (1) and (11).

The isentropic efficiencies were calculated again while each loss parameter (a_{ref}, b_{ref}, c_{ref}) was separately halved to simulate a compressor tailoring by reducing losses of a specific loss mechanism (Figure 5 top). By halving each loss, an extreme case is considered since this loss reduction would be very ambitious to achieve in practice. The displacement required for a heating power of 3 kW is iterated again for each set of a_{scale}, b_{scale} and c_{ref}. As expected, every loss reduction increases isentropic efficiencies compared to the basic case (Figure 5 top). Halving the friction losses (squares) has the greatest effect, while halving electric (stars) losses has the smallest effect. For both cases, the efficiencies are nearly parallel shifted to higher values. Changes in the refrigerant ranking (order by efficiency) are not observed.

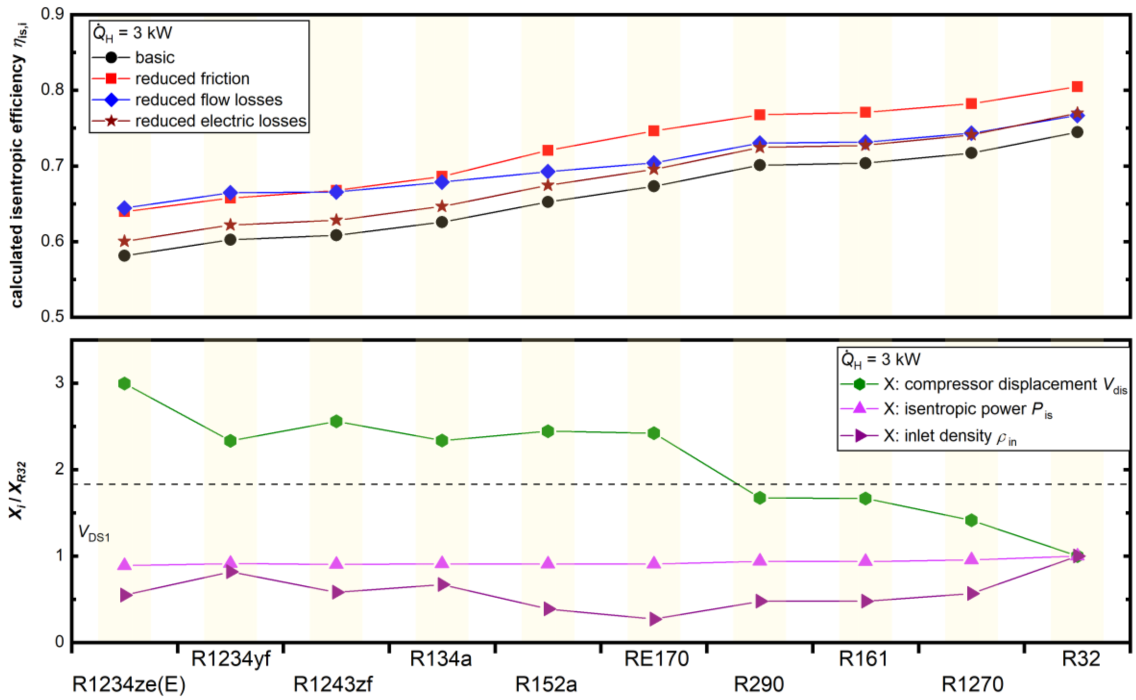

Figure 5:
Top: Isentropic efficiencies resulting from the combined compressor and heat pump model for various refrigerants. Different cases for a, b, c. Basic (circles): a_{scale}, b_{scale}, c_{ref}; reduced friction (squares): a = 0.5·a_{scale}, b_{scale}, c_{ref}; reduced flow losses (diamonds): b = 0.5·b_{scale}, a_{scale}, c_{ref}; reduced electric losses (stars): c = 0.5·c_{ref}, a_{scale}, b_{scale}.
Bottom: Required compressor displacement, isentropic power and inlet density for a heating power of $\dot{Q}_H$ = 3 kW. Calculated using a_{scale}, b_{scale}, c_{ref}.

Halving the flow losses affects the isentropic efficiency of refrigerants differently (*Figure 5* top, diamonds). Refrigerants that require a larger compressor displacement benefit more from a flow loss reduction than refrigerants which require a small displacement. However, changes in the ranking only occur for neighboring refrigerants with already very similar efficiencies. For example, R1243zf benefits less from a flow loss reduction than R1234yf, resulting in a change in the ranking between R1234yf and R1243zf.

5. Conclusions

The performance of heat pumps and cooling processes depends on the refrigerant used. Still, refrigerant selection studies often assume constant compressor efficiencies for all refrigerants. Here, we investigated whether similar isentropic compressor efficiencies could be achieved for all refrigerants by tailoring the compressor design to each refrigerant. Our work shows:

- isentropic efficiencies are highly refrigerant-dependent: the most efficient refrigerant (R32) has a 27% higher efficiency than the refrigerant with the lowest efficiency (R1234ze(E)),
- tailoring the compressor to refrigerants by reducing losses can change the ranking of refrigerants with very similar performance and slightly reduce the overall range of efficiencies between all refrigerants,
- even when losses are halved, large differences in isentropic compressor efficiency remain
- realistic measures for tailoring compressors to refrigerants do not have the potential to overcome differences in efficiency caused by refrigerant properties.
- equal isentropic compressor efficiencies for all refrigerants or changes in the fundamental refrigerant ranking are not achievable for reciprocating piston compressors.

NOMENCLATURE

D	cylinder bore	(m)
f	frequency	(Hz, rpm)
h	specific enthalpy	($kJ \cdot kg^{-1}$)
H	stroke	(m)
P	power	(W)
p	pressure	(Pa)
T	temperature	(°C)
V	volume	(m^3)
$\dot{V}$	volume flow rate	($m^3 \cdot s^{-1}$)
w	specific work	($kJ \cdot kg^{-1}$)
Subscript		
cond	condensation	
cyl	cylinder	
dis	displacement	
elec	electric	
evap	evaporation	
in	compressor inlet	
is	isentropic	
mech	mechanical	
out	compressor outlet	
ref	reference, related to data set 1 (DS1)	
sat	saturated	
scale	scaled referred to DS1	

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ACKNOWLEDGEMENT

This research work was carried out with the support of the Swiss Federal Office of Energy (SFOE) as part of the SWEET (SWiss Energy research for the Energy Transition) project DeCarbCH. The authors bear sole responsibility for the conclusions and the results.